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COGAS Propulsion for LNG Ships

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Abstract: Propulsion of liquefied natural gas (LNG) ships is undergoing significant change. The traditional steam plant is losing favor because of its low cycle efficiency. Medium-speed diesel-electric and slow-speed diesel-mechanical drive ships are in service, and more are being built. Another attractive alternative is combined gas and steam turbine (COGAS) drive. This approach offers significant advantages over steam and diesel propulsion. This paper presents the case for the COGAS cycle.
Keywords: liquefied natural gas (LNG); combined gas and steam turbine(COGAS); gas turbine; economics; emissions
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1 Introduction

As the LNG transportation business looks at alternatives to the traditional geared steam turbine, a variety of approaches is under consideration. The choice among these alternatives should be based in large part on economics and emissions concerns. In the future, there may be a direct economic penalty, in the form of taxes, for the emissions a ship produces.

The steam plant has remained the propulsion system of choice for LNG ships even as owners of other ship types opted for more efficient approaches. A major reason for this is the ease of dealing with the boil-off gas (BOG), which is burned in the propulsion boilers. Continuing increases in fuel costs, and the desire to deliver all the cargo that was loaded, have recently resulted in the consideration of other propulsion systems. Both medium-speed diesel-electric and slow-speed diesel-mechanical drive ships are in service and more are under construction.

With the medium-speed approach, the BOG can be burned in the engine or reliquefied and returned to the cargo tanks. Ships using both of these approaches are already in service. The same choice exists for the slow-speed approach, but only the reliquefaction alternative has actually been built. Burning the BOG in a two-stroke engine requires the injection of natural gas at pressures of 250 to 300 bar. Such systems are under detailed study, but so far concern for the safety of this high-pressure gas has deterred potential ship owners from adopting this approach.

The world-wide order book of LNG ships to be delivered in 2008 was about 43% steam, 40% slow-speed diesel with reliquefaction, and 17% dual-fuel diesel-electric. Going forward in time, the market share of steam is steady while dual-fuel diesel-electric propulsion grows at the expense of slow-speed diesel with reliquefaction.

The simple gas turbine suffers from relatively low cycle efficiency, but the combination of a gas turbine, a waste-heat boiler, and a steam turbine offers a cycle efficiency of up to 55%, which is slightly higher than that of a slow-speed diesel. This value represents the gross efficiency, that is, the power produced by the gas turbine system plus the power produced by the steam turbine system divided by the rate of heat input. (The gross efficiency does not subtract out the parasitic loads such as electrical power for machinery and accommodations.) A COGAS propulsion system is also smaller and lighter than a slow-speed diesel system of the same power output. Thus it appears that the combined gas and steam turbine (COGAS) alternative should be given serious consideration.

This paper makes the case for COGAS propulsion of LNG ships. Economic and emissions considerations are analyzed to demonstrate the attractiveness of COGAS.

2 COGAS history

COGAS plants have been in use ashore since the late 1970s. As of 1996 the shore-side electrical generating capacity of COGAS plants totaled 85 GW (Kelhofer, 1999). A recent news story in Diesel & Gas Turbine Worldwide (Futtsu, 2007) reports that Tokyo Electric Power Company's Futtsu Thermal Power Station will have COGAS systems generating more than 5 GW upon completion of the current expansion program in 2010.

Consideration of COGAS propulsion for ships was proposed by Mills (1977) and Brady (1981). An M.S. thesis at the Naval Postgraduate School (Combs 1979) provides an extensive analysis of a proposed COGAS system for a naval destroyer. An extensive study of COGAS feasibility was conducted for the US Maritime Administration by George G.

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Sharp, Inc. (Giblon 1979). About the same time LNG was considered among several propulsion alternatives for LNG ships (Howard, 1982). Howard dismissed COGAS early in the analysis, but a detailed discussion (Scott 1982) published with Howard's paper shows COGAS to be very attractive. In response to Scott's discussion, the paper's authors stated, "Economically, we do not believe the COGAS plant would be attractive compared with the diesel engine with reliquefaction except when the fuel gas price is more than 10% below the heavy fuel oil price on a heat value basis." The following section of the present paper shows that the price of natural gas in late 2010 was roughly one third of the price of heavy fuel. A detailed analysis of the U. S. Navy Rankine Cycle Energy Recovery (RACER) proposal (Halkola, 1983) also shows COGAS to be attractive. In a presentation at the Chesapeake Marine Engineering Symposium, Harbach (1988) analyzed and optimized steam systems that used waste heat from both diesel engines and gas turbines. More recently, a presentation (McKesson 2002) at the Seatrade Miami conference about Celebrity Cruise's *Millennium* and *Infinity* as well as Royal Caribbean's *Radiance of the Seas*, reported, "General Electric's original estimates claimed that as many as 50 additional passenger cabins could be realized as a result of installing a COGAS plant in the original engineroom space designed for a diesel plant. In both *Millennium* and *Radiance of the Seas*, the designers did, in fact, find this much space, and the cabins were added."

3 Fuel price issues

The choice between burning and reliquefying the BOG should be based primarily on the comparative cost of a BTU of heat from each alternative. In late 2010, the cost of natural gas on the NY Mercantile Exchange was about $4.25 USD per million BTU. This is written as $4.25/MMBTU. At the same time the cost of IFO 380 in Houston was 488.50 USD per metric ton. For an IFO heating value of 18 500 BTU/lb, that equates to $12.00/MMBTU, about three times the cost of natural gas.

The prices of HFO and natural gas are both very volatile. Prices vary from port to port, and there are sharp peaks and valleys as a function of time. LNG ships are often chartered for 20 years. Forecasting the price relationship between IFO 380 and natural gas over that time period is impossible to do with confidence. Slow-speed diesel engines with reliquefaction are the propulsion systems of choice for the Q-Flex and Q-Max ships being built for the Qatar project. This would appear to indicate that the owners believe that the price of natural gas will surpass the price of IFO 380 in the foreseeable future. However, if the current price relationship holds true into the future, it appears clear that burning the BOG is economically attractive.

4 Emissions Issues

There are four principal emissions produced by main engines: oxides of nitrogen (NO_x), oxides of sulfur (SO_x), carbon dioxide (CO_2), and particulates. These four substances are discussed briefly below.

Because there is nitrogen in atmospheric air, the production of NO_x is unavoidable. However, the quantity of NO_x is a function of the fuel and a strong function of the combustion temperature. Higher temperatures result in more NO_x, and combustion of natural gas produces less NO_x than combustion of HFO. Peak temperatures in diesel engines are significantly higher than in gas turbines, so they produce more NO_x. Production of SO_x is governed by the sulfur content of the fuel. HFO typically contains significant sulfur, but low sulfur HFO is available at higher cost. Natural gas contains only a trace of sulfur, so any engine burning natural gas produces very little SO_x.

CO_2 production depends primarily on the ratio of hydrogen to carbon in the fuel and the cycle efficiency of the propulsion system. Liquid fuels such as HFO and MDO have a ratio of about 2:1, while methane, the dominant constituent in natural gas has a ratio of 4:1. As a result, combustion of liquid fuels produces significantly more CO_2 than combustion of natural gas. Particulates are primarily carbon particles. They result from localized areas where the air-to-fuel ratio is too low. The intermittent nature of diesel engine combustion is inclined to produce more particulates than the steady combustion in a gas turbine.

The numbers in Table 1 below are based on kW·h of heat released, so cycle efficiency must be factored in when estimating the emissions per kW·h of mechanical energy output. If two plants produce the same amount of emissions per kW·h of heat, the one with the higher cycle efficiency produces lower emissions per kW·h of mechanical energy output.

Table 1 Emissions comparison [ConocoPhillips]

Propulsion system	NO_x (g/kW·h)	SO_x (g/kW·h)	CO_2 (g/kW·h)	Particulates (g/kW·h)
Steam turbine (50% BOG, 50% HFO)	1	11	950	2.5
Slow-speed diese l (HFO with 2% sulfur)	17	7.7	580	0.5
Medium-speed diesel (BOG)	1.3	0.05	445	0.05
Medium-speed diesel (HFO with 2% sulfur)	12	7.7	612	0.4
Medium-speed diesel (MDO)	12.5	2	620	0.04
COGAS (BOG)	2.5	0	480	0.01

As Table 1 shows, a COGAS plant is at or near the lowest value for each of the four emissions. Since COGAS plants will be shown to have the highest cycle efficiency of the various alternatives, they look very attractive in terms of emissions production.

5 Space and weight issues

As mentioned earlier, gas turbines are exceptionally small and light for the power they produce. While the addition of the steam system increases space and weight requirements, the combined system is expected to provide considerable savings in these areas compared to diesel propulsion. McKesson (2002) presents dimensions and weights for several gas turbines. The Rolls-Royce TRENT engine produces 47.5 MW from a unit that weighs 26 t and has a footprint of 44 m^2. By comparison a 6-cylinder K98MC slow-speed diesel engine from MAN B&W produces 46.7 MW from a unit that weighs 1152 t and has a footprint of about 60 m^2. Of course the steam system will add to the space and weight requirements of a COGAS plant. Halkola (1983) provides an estimate of the weight of the steam machinery. For a steam turbine power of 7500 hp (1 kW=1.341 hp), the estimated weight of the steam machinery is about 27 t. Writing about a steam system that would produce 8700 hp, Mattson (1983) states, "… although a single RACER system will weigh 40 to 60 t, there would be a reduction in mission fuel weight of several times this weight." In his thesis Combs (1979) estimates that a waste heat boiler for a gas turbine operating at 18 000 horsepower might have dimensions of 12 feet (1ft=0.3048m) by 12 feet perpendicular to the flow direction and 7 feet parallel to the flow direction. Abbott (1974) predicted that the entire waste heat boiler would weigh 59 000 pounds (1 pound=0.4536 kg) when dry and 65 000 pounds when wet. He predicted dimensions as follows: height 5.5 feet, length 15 feet, width 11.5 feet. Abbott's prediction of the total machinery weight, including the gas turbine, would be 575 t.

In addition to machinery, fuel storage for engines burning liquid fuel contributes to the overall size and displacement of the vessel. The Super-Flex ships designed for the Qatar project will carry 9 800 t of HFO and 600 t of MDO (Noble, 2007). The corresponding volume would be more than 10 000 m^3. These ships are designed to carry 228 500 m^3 of LNG.

With COGAS propulsion, the BOG provides the primary fuel. A modest amount of liquid fuel would probably be carried for emergencies. It is not clear how much the cargo capacity could be increased if the need to carry HFO were eliminated. However, it is interesting to note that if the boil-off rate is 0.11% per day, that amounts to about 250 m^3 of LNG per day. The volume of liquid fuel carried in the Super-Flex ships corresponds to about 40 days of boil off.

6 Maintenance issues

Gas turbines are noted for requiring relatively little maintenance. McKesson (2002) states, "As opposed to changing or repairing major components on a set schedule, which is normally the case with diesel engines, repairs to the LM2500+ [gas turbine] sets are carried out based on conditions observed during regular borescope inspections. These are normally done approximately once every 2 500 hours." He further states, "The last borescope inspection on *Millennium* was carried out in January 2001. The service engineer stated that internal components still looked like new after 5 000 hours of operation. At that rate, it is the opinion of GE experts that the predicted 15 000 hour hot-section repair interval will be easily passed."

The comments above pertain only to gas turbine machinery. However, the steam system in a COGAS system is also expected to require little maintenance. Mills (1977) points out, "… the combined-cycle steam generator experiences much more benevolent gas-side temperatures and none of the conventional boiler's problems associated with flame, oil, or ash impingement …" Mattson (1983) discussed design issues that would significantly influence the reliability and maintainability of the US Navy's proposed RACER system. He goes on to predict that the steam machinery will require "… less than 50 man-hours per week maintenance." This estimate applies to a naval environment where maintenance is time-based rather than performance-based.

7 COGAS System description

COGAS systems can have one boiler or multiple boilers operating at different pressures. The multiple boiler case is more efficient than the single boiler case, but it comes at the cost of additional space and weight. Such systems are to be found ashore, but for purposes of the current analysis, only the single boiler case is considered for a shipboard application. Fig.1 shows the schematic diagram of a single boiler COGAS system with a deaerating feed tank (DFT). State points in the gas cycle are designated by numbers, and state points in the steam cycle are designated by letters. The gas cycle is a simple one with a compressor, a heater, and a turbine. The first few stages of the turbine drive the compressor, and the remaining stages, mounted on a separate shaft, drive the external load. Although it is not common practice, it is possible to separate the turbine that drives the compressor from the turbine that drives the external load and place a reheater between the turbines. This device can raise the temperature back to the value leaving the first heater. The result is an increase in cycle efficiency of about 7%. No reheater is considered in the analysis below.

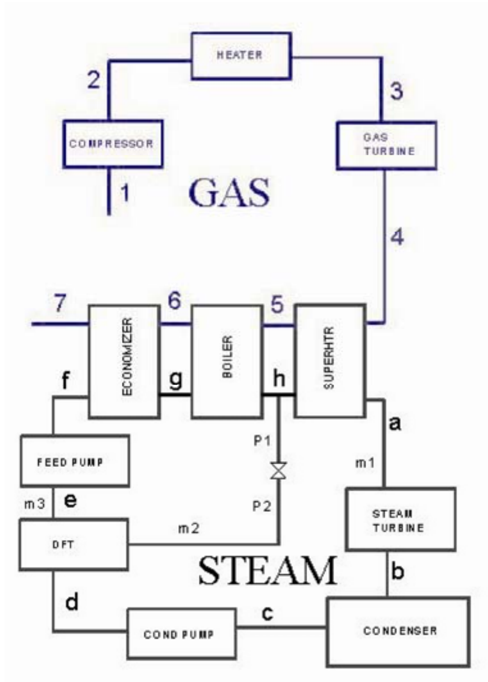

Fig.1 COGAS schematic

Leaving the gas turbine, the exhaust gas passes through three heat exchangers: the superheater, the boiler, and the economizer. (In actual practice, these may be combined into a single device, but for purposes of analysis, they are considered separately.) Steam leaving the superheater is expanded through a turbine and exhausted to a condenser. A condensate pump moves the water to the DFT, and a feed pump moves it to the economizer. This system is very similar to a conventional steam system. The difference lies in the heat exchangers, because no fuel is burned there.

Fig.2 shows a simplified arrangement of a single pressure COGAS system. Note that the intake ductwork is not shown here.

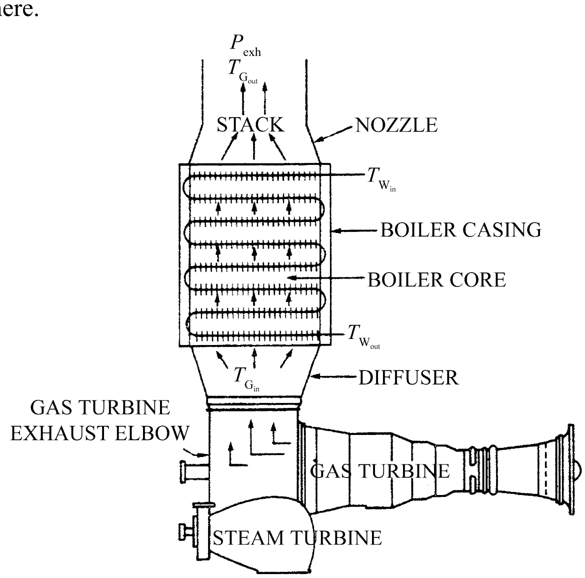

Fig.2 COGAS arrangement

In the following analysis reasonable efficiencies are assumed for the gas compressor, the gas turbine, and the steam turbine. Although the pumps in the steam cycle are not isentropic, it is assumed that they are. This assumption simplifies the analysis, and the error introduced is insignificant. The working fluid in the gas turbine system is analyzed on an air-standard basis, i.e., it is assumed to be pure air behaving as an ideal gas with variable specific heat.

All components are assumed to be well insulated so that heat transfer to the environment, except in the condenser, is negligible. Energy is released to the environment at state 7, and this is accounted for in the analysis.

Steam for the DFT is taken from the boiler outlet where it is saturated vapor. However, this steam is throttled to lower pressure for use in the DFT, thereby reducing the required strength of this component. The throttling process does not change the enthalpy of the steam, but the pressure decrease renders the steam superheated.

8 The computer model

Mathcad computer software has been employed to create a complete analytical model of the system shown in Fig.1. This model includes access to the thermodynamic properties of air and of steam and water. It can be used to quickly and easily explore the effect of any design variable. It computes the values of heat and work in each of the system components and the thermodynamic state at each point (1-7 and a-h) in the diagram.

In addition to values of enthalpy, heat, and work, the computer model calculates cycle efficiency and specific fuel consumption. It performs a basic economic analysis that gives a value for the present worth of fuel savings resulting from the addition of the steam system to the gas turbine system.

All of the numerical results presented in this paper were calculated using the Mathcad computer model. Additional computer models have been created to analyze the multiple boiler case and the reheater case, but these models are not involved in the following analysis.

9 Heat exchanger pinch points

Typical temperature profiles of the air and water/steam as they flow through the heat exchangers are shown in Fig.3. Special attention must be given to the point where the water leaves the economizer and the point where the steam leaves the superheater. These points, where the temperature difference between air and water/steam are smallest, are called the pinch points.

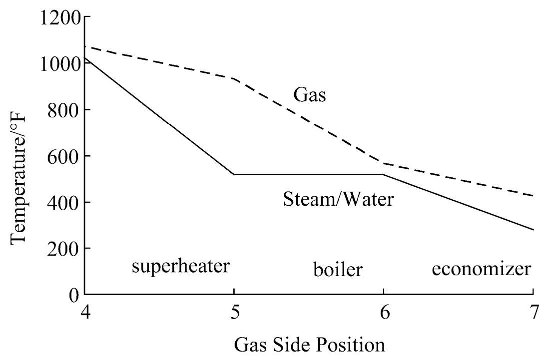

Fig.3 Typical pinch point diagram

Although the temperature differences at the heat exchanger pinch points are under the designer's control, the choice has significant consequences. The temperature difference and the required amount of heat transfer surface area are inversely proportional. A smaller temperature difference requires more heat transfer surface and a larger, heavier, and more expensive heat exchanger. However, a smaller temperature difference means higher cycle efficiency, because it represents more thermal energy removed from the gas turbine exhaust. In the analysis that follows, the temperature difference at each of the pinch points is taken as 50 ºF [℃ = (ºF−32) / 1.8].

10 Design variables

There are many variables in the design of a COGAS system. Some of these are under the designer's control, and some are determined by external constraints. As mentioned above, the pinch point temperature differences are under the designer's control. Other variables under the designer's control include boiler operating pressure, gas compressor pressure ratio, gas turbine inlet temperature, and DFT operating pressure. The design process involves finding the values of these parameters that maximize the cycle efficiency while keeping in mind constraints on space, weight, and cost.Although values of atmospheric pressure and temperature and condenser vacuum vary, they are not under the designer's control. For purposes of this analysis, inlet air pressure is taken to be 14.5 psia (1 MPa=145.037 25 psia), and condenser vacuum is taken to be 0.75 psia. The inlet air temperature at point 1 is taken to be 80 ºF.

Table 2 contains the baseline values of variables under the designer's control.

Table 2 Baseline values of design variables

Boiler pressure /psia	800
DFT pressure /psia	50
Compressor pressure ratio	15
Gas turbine inlet temperature /ºF	2200
Compressor inlet temperature /ºF	80
Steam turbine efficiency /%	85
Gas compressor efficiency /%	85
Gas turbine efficiency /%	87
Superheater outlet temperature / (ºF below gas turbine exhaust)	50

11 Baseline results

Calculations were performed for the production of a net power output of 25 000 horsepower. The power inputs for the pumps in the steam cycle have been ignored, because they are very small. Major power inputs and outputs are presented in Table 3.

Table 3 Power Output

Component	Power / hp
Gas Compressor	-22 820
Gas Turbine	40 440
Steam Turbine	7 380
Net Output	25 000

Heat input to the gas heater is 127.3 million BTU/hour. The calculated cycle efficiency is 50.0%. This value is similar to the efficiency of a slow-speed diesel engine. However, because the BOG is being burned at relatively low pressure, neither a high-pressure BOG compressor nor a reliquefaction plant is needed.

Required flowrates of steam, air, and BOG for the baseline case are presented in Table 4.

Table 4 Required Flowrates

	Flowrate / (lb/hr)
Steam to steam turbine	38 340
Air	328 500
BOG	6 968

12 Economic analysis

If a steam system is to be added to a gas turbine system, it must be demonstrated that the resulting fuel savings justify the cost of the steam equipment. For purposes of this analysis, the value of natural gas is assumed to be $4.25/MMBTU. The minimum attractive rate of return on invested capital is assumed to be 10%, and an analysis period of 20 years is used. Based on these assumptions, fuel savings that result from addition of the steam machinery are about $1.6 million per year. The present worth of these savings is about $13.5 million. If the steam machinery can be purchased and installed for less than this amount, addition of the steam system is justified. In late 2010 natural gas prices are at historic lows. If the value of natural gas rises in the future, the present worth of savings will, of course, increase.

13 Variation of design parameters

It is useful to explore the effect of some of the design parameters on the cycle efficiency. In particular, the effects of boiler pressure, compressor pressure ratio, and gas turbine inlet temperature are demonstrated below. This analysis is facilitated by the Mathcad computer model. The results below are simply a sample of what can be done with this model. Many other studies can be conducted quickly and easily.

When the boiler pressure is lowered, more steam is produced, but the energy content of that steam is lower. The optimum steam pressure is not obvious. Fig.4 below shows how the cycle efficiency changes with boiler pressure. All other design variables remain at the values designated in Table 2.

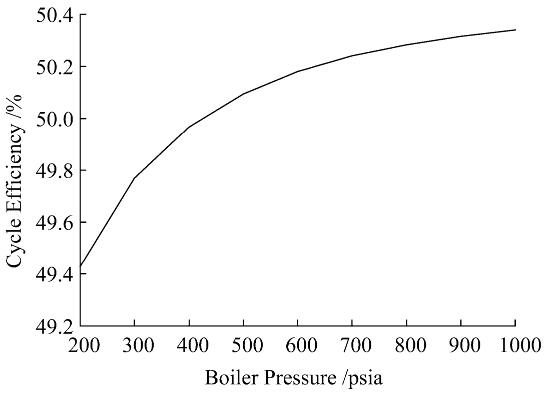

Fig.4 Effect of boiler pressure on COGAS efficiency

Within the range of pressures explored, there is no relative maximum in cycle efficiency. However, the gain in cycle efficiency becomes progressively smaller at high pressures. Increasing boiler pressure implies increasing weight and cost. The optimum boiler pressure is therefore an economic decision.

For purposes of the following analyses, a total required power of 25 000 horsepower is assumed. Fig.5 shows how the power is distributed between the gas turbine and the steam turbine for a range of boiler operating pressures.

Gas turbine power is represented by the solid line and is read along the left axis of Fig.5. Steam turbine power is represented by the dashed line and is read along the right axis. Increasing boiler pressure results in more power from the steam turbine. Since the total power is to remain at 25 000 horsepower, the power required from the gas turbine decreases as the boiler pressure increases. Over this range of boiler pressures, the gas turbine power decreases by 1.8% while the steam power increases by 4.5%.

Fig.6 shows how the flowrates of air and steam vary with boiler pressure. The steam flowrate being shown here is the flowrate through the steam turbine. This is slightly less than the flowrate of steam leaving the boiler, because a small amount of that steam is fed to the DFT.

Air flow is represented by the solid line and is read along the left axis, while steam flow is represented by the dashed line and is read along the right axis. Both flowrates decrease as boiler pressure increases. Over the range of 200 psia to 1000 psia, the air flowrate changes by only 1.8%, but the steam flowrate changes by 10%. Although these effects are fairly small, they may have consequences in terms of pipe and duct sizes.

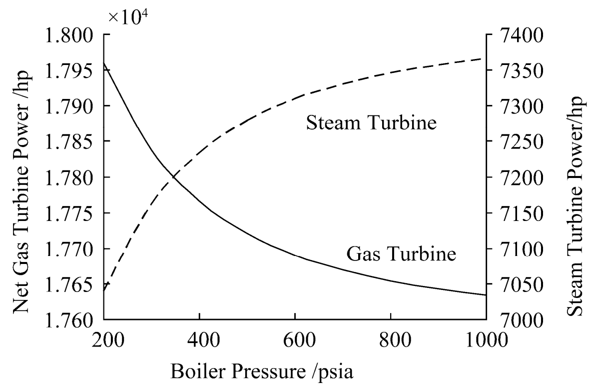

Fig.5 Power distribution between gas and steam turbines

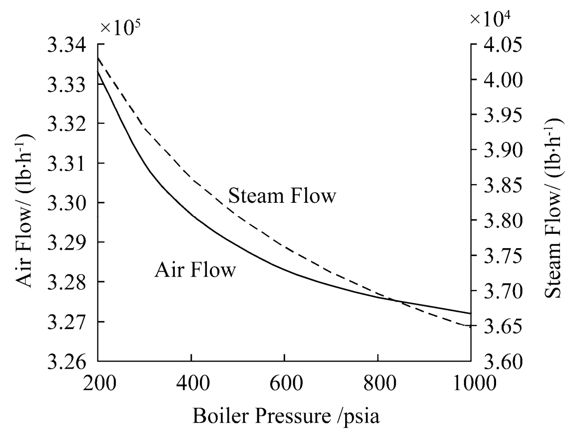

Fig.6 Air and steam flowrates for 25 000 horsepower

Fig.7 shows the variation in required BOG flow as a function of boiler pressure. The inherent BOG flowrate is not under the control of the propulsion system designer. Rather, it is a consequence of ship size and LNG containment performance. If the inherent BOG flow is insufficient, heat can be applied to the cargo tanks to increase it. It is not likely that BOG flow will be more than required when operating at or near full power.

However, there will be excess BOG during maneuvering and at the dock. Provision must be made for handling this excess. While the details are beyond the scope of this paper, the excess BOG might be reliquefied or flared off (burned).

The required BOG flow decreases by about 1.8% as the boiler pressure rises from 200 psia to 1000 psia.

Fig.8 shows how the cycle efficiency varies as a function of the pressure ratio in the gas turbine cycle. In this case, there is a distinct maximum in cycle efficiency at a pressure ratio of about 12.5.

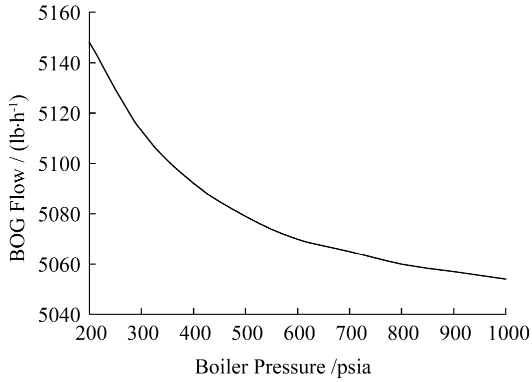

Fig.7 BOG flow to produce 25 000 horsepower for various boiler pressures

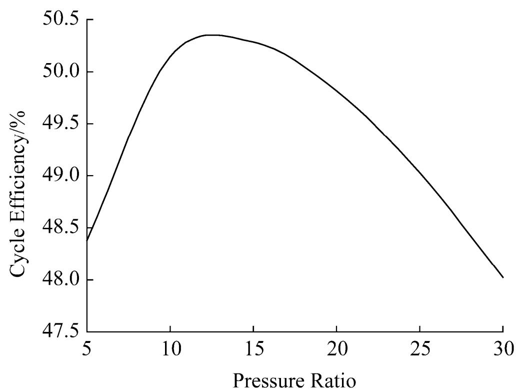

Fig.8 Effect of gas turbine pressure ratio on COGAS efficiency

Fig.4 and Fig.8 show that boiler pressure and gas compressor pressure ratio have modest effects on cycle efficiency. However, the gas turbine inlet temperature has a much larger effect. In this analysis, the superheater outlet temperature (Point a in Fig.1) is set at 50ºF below the gas turbine exhaust temperature (Point 4 in Fig.1), which rises as the gas turbine inlet temperature rises. Fig.9 shows that with inlet temperature approaching 3000 ºF, the cycle efficiency rises toward 58%.

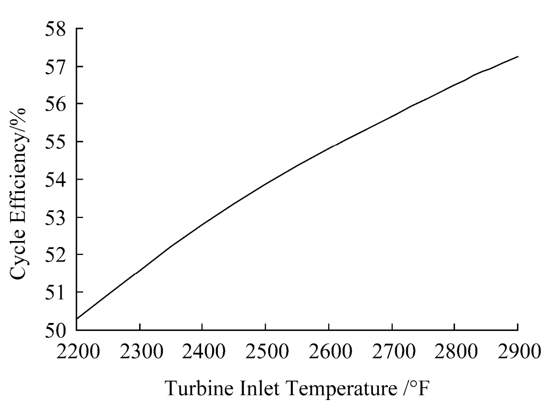

Fig.9 Effect of gas turbine inlet temperature on COGAS efficiency

Gas temperatures this high are possible, but they can not be achieved with conventional turbine blades. Instead of solid blades, high-performance gas turbines must use blades with cooling passages. There are three possible approaches. One method involves a small amount of air from the compressor that is fed directly to blade passages that lead to holes in the surfaces of the blades. The air ejected through these holes creates a layer of relatively cool air between the hot gas from the heater and the blade metal.

A second cooling method is called transpiration cooling. With this approach, liquid water is pumped to the cooling passages and allowed to boil off through the openings in the blade surface.

The third cooling method cools the blades with some of the exhaust steam from the high pressure turbine and uses the blade cooling process to reheat the steam before sending it to the intermediate pressure turbine. This method is described in detail by Najjar (2004).

Gas turbine engines being used in military aircraft have achieved heater outlet temperatures as high as 3600 F (Langston 2007). These engines obviously require special materials and extensive maintenance. Temperatures this high are probably not feasible for ship propulsion, but temperatures a few hundred degrees above 2200 F may be feasible.

14 Other system variables

The temperature of the air entering the gas compressor can vary significantly from place to place and from season to season, but it is not under the designer's control. Air temperatures in Qatar in summer may reach 110 ºF, while air temperatures in the Chesapeake Bay in winter may be 30 ºF or lower. The propulsion system must be designed with this temperature variation in mind. If the efficiencies of the gas compressor and turbine are held constant, increasing inlet air temperature causes a very small decrease in cycle efficiency. At constant component efficiency, increasing inlet air temperature also causes a requirement for a higher mass flowrate of air, and this air has decreased density. Higher mass flowrate plus lower density results in a significant increase in required volumetric flowrate as shown in Fig.10. It is beyond the scope of this paper to address the change in component efficiencies of the gas compressor and turbine under these conditions. Suffice it to say that the assumption of constant component efficiencies is not reasonable. In reality these component efficiencies will decrease as volumetric flowrate increases, and that will cause a decrease in cycle efficiency.

At constant component efficiency, the volumetric flowrate of air required to produce a total output power of 25 000 hp rises from 1154 ft^3/s at an inlet temperature of 60 ºF to 1424 ft^3/s at 120 ºF, an increase of about 21%.

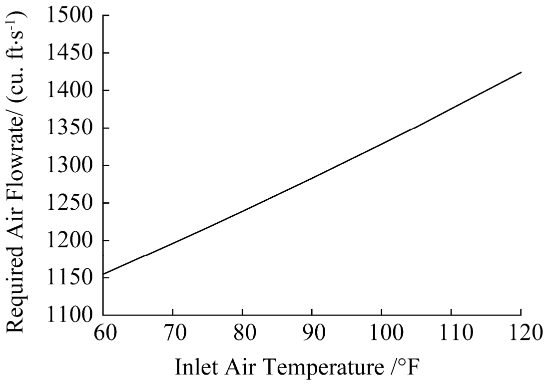

Fig.10 Effect of gas compressor inlet temperature on volumetric flowrate to produce 25 000 horsepower

15 Dynamic performance

This paper attempts to demonstrate that COGAS propulsion is an attractive alternative for the propulsion of LNG ships, but the analysis is limited to steady-state operation at full power. Clearly, the system must perform satisfactorily at reduced power and during maneuvering. Abbott (1977) described a detailed analysis of the dynamic performance of a COGAS system and concluded "… it has been shown that a COGAS plant can be designed with an uncomplicated control system capable of stable operation in the steady-state condition and throughout the full range of extreme ship maneuvers."A recent doctoral thesis (Jefferson 2006) at the University of Newcastle upon Tyne presents results of a computer simulation of the dynamic behavior of such a system. The conclusion of that work is that acceptable dynamic behavior can be achieved.

16 Conclusions

This paper demonstrates that the cycle efficiency of COGAS propulsion is comparable to that of a slow-speed diesel engine. With a slow-speed diesel burning BOG, a high-pressure compressor is necessary, while COGAS requires only a low-pressure compressor. Thus even at the same cycle efficiency, COGAS is cheaper to operate. The paper further demonstrates that for 25 000 horsepower total output, the present worth of fuel savings associated with adding the steam system to the gas turbine is about $13.5 million at 10% for 20 years, based on the current price of natural gas. If the cost of the steam machinery is less than this amount, addition of the steam equipment is justified.

It should be noted that with either slow- or medium-speed diesel engines, a wide range of powers is available. The design power can be incremented in relatively small steps. Furthermore, diesel engine efficiency is nearly constant from full power down to approximately 50% power. In contrast, gas turbines are available in only a few discrete power ratings, and their efficiency deteriorates rapidly as a given turbine's power is reduced, as shown in Fig.11 taken from Brady (1981).

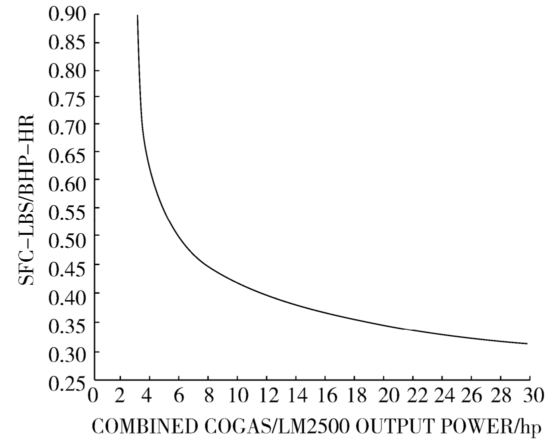

Fig.11 Effect of reduced power operation on COGAS SFC

Reduced efficiency at reduced power may be a blessing in disguise for an LNG ship where the rate of BOG production is independent of ship speed. It means that nearly the same amount of BOG must be burned at reduced power as at full power.

Based on the analysis above, it appears that COGAS propulsion is competitive with other approaches to the propulsion of LNG ships. In view of this, COGAS propulsion deserves further detailed study.

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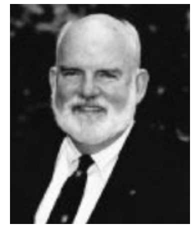

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